



Study and application of a regenerative Stirling cogeneration device based on biomass combustion



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HIGHLIGHTS

- The recovery of energy content of combustion flue gas in biogas-fed Stirling co-generators is studied.
- A spiral gas–gas heat exchanger is used.
- Electric efficiency can be raised up to 22.5%.
- The system is applied to a residential case study following optimal management criteria.
- The extra-cost of the recuperator is paid back in less than two years.

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ABSTRACT

Fuelling micro-combined heat and power (micro-CHP) devices using renewable sources, such as biogas, can enhance the energy and environmental benefits associated with such devices. This paper presents a solution for increasing the electric efficiency of biogas-fed Stirling co-generators by recuperating the energy content of the combustion flue gas. When economically and energetically convenient for micro-CHP operation, the exhausts can be used to pre-heat fresh combustion air. It was assessed that, due to the introduction of a spiral gas–gas heat exchanger, whose main design parameters were identified, the electric efficiency of the Stirling unit can be raised to up to 22.5%. To determine the advantages of applying the system analysed over a traditional Stirling device, a specific algorithm for the optimal management of the micro-CHP unit was built and applied to a residential case study. The results demonstrate that the extra cost of the high-efficiency recuperator can be easily recovered in approximately two years of system operation, providing an additional advantage over a standard Stirling device in terms of primary energy consumption and an approximately 6% increase in economic savings.

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1. Introduction

The European Union has defined guidelines for a more efficient and rational use of energy and has identified certain targets for Europe's energy policy: reduce the dependence on imported energy and exposure to volatile fossil fuel prices; reduce the emission of greenhouse gases (GHGs) by using cleaner, locally produced energy; and develop competitive energy markets to stimulate technological innovation.

Combining distributed power generation with heat supply (CHP) in households by means of small micro-cogeneration units is a novel way to maximise energy efficiency, secure the supply of

energy and reduce CO₂ emissions. Both the EU and national governments encourage the widespread use of micro-CHP systems, which would help to meet the targets for carbon emissions [1]. The advantage of using distributed generation could be enhanced if micro-CHP units are fuelled by alternative fuels or renewable resources [2], providing an important emission cutting to the building sector [3].

Among the alternative fuels that are available, biogas is one of the most valuable for the local production of energy because its methane content makes it possible to use this fuel in several energy-conversion devices; thus, it can be used directly for heating and generating electricity [4].

Biogas production by anaerobic digestion (AD) is one way to meet the targets identified by the European Union. Over 80% of the estimated potential of biogas production derives from agricultural resources and from municipal organic waste [5]. When

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compared with other renewable energy sources such as solar or wind power, biogas is more versatile because its production is steadier and it is easy to store; as a consequence, its use is independent of geographical location and ambient conditions, which is a key advantage in the overall management of the electric grid [6].

The production of biogas in small-scale plants fed by household organic waste has already been studied and demonstrated [7] both in industrialised and developing countries. For example, biogas usage was analysed: to feed cattle farms in Ontario [8] or a brewery in UK [9]; as an energy resource from crops in Sweden [10]; and by anaerobic digestion in India [11]. The wide availability of this fuel would encourage the use of micro-cogeneration systems such as internal combustion engines, Stirling engines, Organic Rankine Cycle engines, micro gas turbines and fuel cells [12].

The use of biogas to fuel internal combustion engines is viable for medium- and large-scale plants that produce 50–2000 kW of power, as reported in Ref. [13], but it is less suitable for small-scale cogeneration. In fact, the variation in the composition of the fuel that could result from using a small-scale digester could be an issue for the operation of internal combustion engines. Biogas could also be used to feed fuel cell devices. Due to its high conversion efficiency, low emission and ability to use multiple fuels, the molten carbonate fuel cell (MCFC) is considered one of the most promising biogas-fuelled devices [14]. However, it is recognised that the related investment costs are still high and some reliability issues must still be resolved; as a consequence, the commercial deployment of these devices has yet to be achieved [15].

Other energy conversion devices that can be fed by biogas are micro gas turbines (MGTs) [16]. The modular MGT modules that are available on the market exhibit an electric power production ranging between 30 and 200 kW and can be incorporated into multiple units to meet user load requirements. Their low combustion temperature results in lower NO_x and CO emissions [17] and good overall efficiency [18]. On the other hand, it is necessary to feed this type of machine with purified compressed biogas, which involves an energy consumption of approximately 10% of the electricity generated [5]. In addition, the size of MGTs is more suitable for medium-scale power generation than for small domestic users.

Organic Rankine Cycle (ORC) technology is used to convert thermal energy from low-temperature heat sources to electricity using high-molecular-mass organic fluids. Thus, they can be fed not with only biogas but also with heat generated by the direct combustion of biomass [19]. The heat-to-electricity conversion efficiency of ORC plants ranges between 5% and 17%, depending on the condensation pressure and the temperature of the heat source. This type of plant is only economically feasible with high annual usage and high system reliability. Therefore, present ORC plant installations using biogas are limited to large-scale plants with thermal power outputs exceeding 300 kW_{th} [20] and where there is low-temperature heat demand. The investment costs are approximately 60% those of an equivalent Stirling engine [21]. Some attempts to run small-scale ORC plants with biomass have been investigated, but although the application is a very interesting solution for small-scale residential applications [22] performance and system costs are still too poor for the diffusion of these devices [23].

Among the electricity generation devices used in small-scale applications, Stirling engines are one of the most promising due to their high overall efficiency, reliability and fuel flexibility [5].

Stirling engines possess interesting technological features that make them suitable for domestic and small-scale applications: quiet operation; good partial load performance; and suitable power-to-heat ratio for applications in residential and commercial buildings [24].

While in traditional internal combustion engines heat is released by the fuel inside the cylinder volume, in Stirling engines, the heat supply can be derived from any external source, which provides the opportunity to use a wide range of energy sources, including fossil fuels and renewables such as solar or biomass. Combustion takes place outside of the engine and thus can be easily controlled; as a consequence, the flue gas contains very few polluting substances and the combustion process is more efficient. Additionally, the number of moving parts is lower compared to that in reciprocating internal combustion engines, resulting in lower wear and maintenance; in addition, Stirling engines exhibit quieter and smoother operation [25].

Some types of biomass-fuelled Stirling engines have been developed, but only a few of them are available on the market and require low-ash-content fuels to avoid fouling issues [26]. High-temperature combustion is required to achieve a significant specific power and efficiency and, at the same time, reduce the problems associated with fouling in the heat exchangers. However, high investment costs and a lack of proven operation are the main obstacles to the distribution and commercial use of biomass-fuelled Stirling engines [27].

Over the past few years, Stirling engines have begun to show good penetration into the market of household micro-CHP units due to the development of a new mechanism, the free-piston Stirling engines, which is particularly suitable for small-scale units [28]. The electric efficiency of these cogeneration units is in the range of 15%, but the overall efficiency can be as high as 90% because the combustion flue gases, after feeding the Stirling engine, still retain a high temperature and energy content and can be used for cogeneration purposes. Currently, most of the Stirling engines available on the market are embedded inside traditional boilers and can be used for household micro-cogeneration fed by natural gas [29]. Some of these engines have also been designed to be fed with alternative fuels such as biogas without requiring significant modifications to the fuel-feeding device because biogas has a high content of methane; this design allows engines to reach higher combustion temperatures, compared with those required for direct biomass combustion, and to avoid the formation of ash and fouling.

In this work, we propose a solution for increasing the electric efficiency of biogas-fed Stirling co-generators by recuperating the energy content of the combustion flue gas. When convenient, the exhaust can be used to pre-heat fresh combustion air. A specific design for a gas–gas heat exchanger is proposed to use the heat of the exhaust and reduce the fuel flow rate required to feed the engine.

Section 2 describes the system analysed, identifying the main design parameters of the recuperator; Section 3 describes the optimal energy dispatch algorithm, which is able to determine, for each time step, whether to convey the exhaust gases to the recuperator based on the energy loads of the end user and fuel and electricity prices. Finally, economic and other aspects besides energy that must be considered to apply the presented design are presented, using a residential application as a case study.

2. System under analysis

The Stirling co-generation boiler described in this work can produce, under nominal conditions, 1 kW of electric power and 3 kW of thermal power by means of the engine's cold-end heat exchanger. The engine is fed with a gas burner that forms a crown around the hot end of the Stirling engine and transfers the thermal power to the working fluid by irradiation and convection. The burner can also be fuelled using other fluids such as LPG or biogas. The nominal working temperature of the engine is 500 °C; thus, the exhaust leaves the combustion chamber at a temperature of 650 °C

Table 1
Stirling cogenerator performance [29].

Property	Value
Engine burner thermal power [W]	6300
Stirling engine nominal electrical power [W]	1000
Electrical efficiency [%]	15.8%
Stirling engine cooling water thermal power [W]	3000
Stirling engine exhaust thermal power [W]	1500
Global thermal efficiency [%]	71.4

and can therefore be used for cogeneration purposes, as described in a paper presented by the manufacturers of the Stirling engine [30]. For this reason, an additional heat exchanger is used to recover the remaining heat for domestic hot water production; therefore, an additional 1.5 kW of thermal power can be recovered.

If the thermal power required by the user is higher than the sum of the thermal power recovered by the engine cooling and that recovered by the exhaust, an auxiliary burner supplies the required amount of thermal energy.

The burner that feeds the Stirling cogenerator has a nominal power of 6.3 kW and uses a gas valve to regulate the amount of combustion natural gas, depending on the fresh air feed. Table 1 reports the performance of the Stirling cogenerator described in this work.

In most applications [31], the cogenerator operates in thermal priority mode to maximise the efficiency of the micro-CHP device; consequently, in the absence of thermal demand, the system is switched off. This control strategy limits the working hours of the system, especially during those periods when the demand for thermal power is limited, while electric energy might be required.

In these cases, it is worth producing electric energy instead of buying it from the grid, only if the electrical efficiency of the system is sufficiently high; consequently, the combined production of heat and power has an overall efficiency that is higher than that of separate energy production. The solution that is proposed in this work optimises the electric performance of the system by using the thermal power that is still available in the exhaust of the engine. As a consequence, the fuel mass flow rate required to feed the engine can be reduced, and thus, the electrical performance of the Stirling cogenerator can be enhanced. The fuel feed device must be modified because the amount of fuel is not proportional to the mass flow rate of the fresh air, as in the original configuration, but it also depends on the energy content of the pre-heated fresh air.

The mass flow rate of the fresh air can be kept constant, while the fuel valve is controlled and retrofitted according to the temperature of the preheated fresh air flow. The aim is to reduce the amount of injected fuel as the temperature of the regenerated fresh air increases.

A schematic of the system is shown in Fig. 1. Under nominal working conditions, the engine's electrical efficiency is 15.8%, but this value can be considerably increased if the discharged heat is recovered.

In this work, a specially designed gas–gas heat exchanger is proposed to use the heat of the exhaust and reduce the fuel flow rate needed to feed the engine. In particular, the recuperator is designed for the biogas feeding of the Stirling engine.

The design of the heat exchanger must be properly evaluated: the amount of heat that can be recovered from the exhaust is limited by the maximum temperature that the fresh air can reach without running into the risk of the auto-ignition of the air-biogas mixture.

The auto-ignition temperature of pure methane is approximately 540 °C. A biogas mixture has a methane content that ranges between 55% and 75%, depending on the digestion process and on

the nature of the biomass that is used in the process [32]. The rest of the mixture is composed of CO₂, which increases the auto-ignition temperature of the biogas to over 650 °C, as reported in Ref. [33] and in Ref. [34]; its flammability range is 7–14% in volume in air. To avoid any risk of auto-ignition of the mixture, one of the design parameters of the recuperative heat exchanger that must be precisely defined is the efficiency of the recuperator in pre-heating fresh air to a maximum temperature of 540 °C, which is the auto-ignition temperature of methane. Because the temperature of the exhaust is known and considering a fresh air temperature of 15 °C, it is possible to evaluate the efficiency of the heat exchanger that is required to preheat fresh combustion air to 540 °C. The relation that defines the efficiency of a heat exchanger is

$$\varepsilon = \frac{T_{c,out} - T_{c,in}}{T_{hot,in} - T_{c,in}} \quad (1)$$

where $T_{c,out}$ is the temperature of the regenerated fresh air, $T_{c,in}$ is the ambient temperature of the fresh air and $T_{hot,in}$ is the temperature of the exhaust entering the heat exchanger. Given this constrain, the maximum recuperator efficiency that can be adopted in order to avoid auto-ignition issues is 83%, which is the value that will be used in the design of the heat exchanger.

2.1. Characteristics of the system

2.1.1. Recuperator design

In addition to the heat exchange properties, the heat recuperator has to meet other specific requirements to suite the aforementioned application:

- compactness and adequate geometry to be fitted over the hot end of the engine;
- capacity to stand very high working temperatures;
- capacity to operate on biogas fuel flue with a range of compositions;
- resistance to fouling and corrosion;
- low pressure drop.

Among the heat exchangers that could be used for this application, the spiral heat exchanger is the most suitable, according to the suggestions reported by Kuppan in Ref. [35]. This type of recuperator meets all of the requirements for these applications and has successfully been adopted in many applications in which a very low mass flow rate is involved but high temperatures and compactness are required. The hot and the cold channels, through which the exhaust and fresh air flow, are wrapped around each other, and heat is transferred to the cold fluid through the interface between the two channels. In this work, an AISI 347 sheet with a thickness of 0.2 mm was used to create the hot and the cold channels of the recuperator.

Swiss-roll heat exchangers have been successfully used to cool high-viscosity and high-density fluids [36]. Another application in which swiss-roll recuperators have been used is in fresh air pre-heating for gas turbines combustors, with the aim of increasing the cycle efficiency. In Refs. [37,38], a wrapped spiral heat exchanger was applied to an industrial turbine, while in Ref. [39], a micro gas turbine was used. This latter application is particularly interesting because the working temperature and the mass flow rate of the fluids are similar to those involved in the proposed recuperator.

To define the geometry of the recuperator, the first step is to evaluate the mass flow rates over the two channels. The flow rate in the hot channel varies as a function of the composition of the fuel because the mass flow rate of the fuel needed to feed the engine decreases with the increasing content of methane in the biogas.

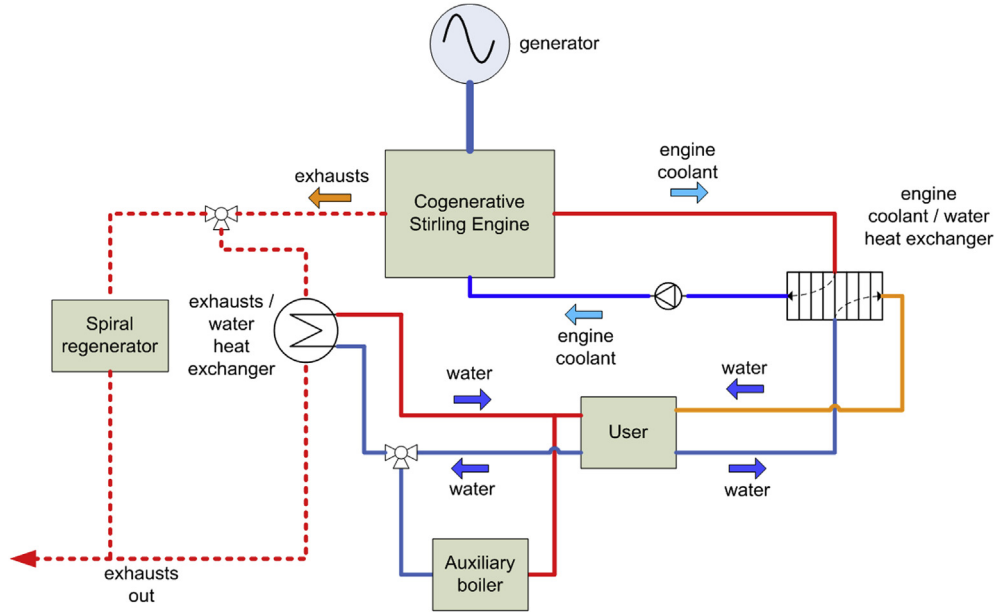


Fig. 1. Schematic of the working principle of the Stirling domestic cogenerator in both regenerative and non-regenerative modes.

For the recuperator design, a volume mixture of 65% methane (with a LHV of 50 MJ/kg) and 35% CO₂ at a temperature of 15 °C is considered. The LHV of the fuel is 20 MJ/kg because CO₂ does not take part in the combustion. Given these conditions, to feed 6.3 kW of power to the Stirling engine, the gas burner requires 0.319 g/s of fuel if a combustion efficiency of 98% is achieved, which corresponds to a methane mass flow rate of 0.129 g/s. The AF ratio of this fuel mixture under stoichiometric conditions is 7.01; considering 40% excess air, as suggested in Ref. [40], the fresh air mass flow rate is 3.13 g/s and the exhaust mass flow rate is 3.45 g/s, as indicated by the following equation:

$$\dot{m} = \frac{P_b}{LHV\eta_{comb}} (AF(1 + ex) + 1) \quad (2)$$

Picon-Nunez et al. [41] defined a procedure for designing a spiral heat exchanger based on the heat exchange equations defined by Minton [42]. The procedure is iterative, and the geometric dimensions of the channels' width and height must be assumed at first; then, the heat exchange performance between the channels is evaluated, and the new total length of the spiral is obtained.

The relation derived by Minton [42] provides the equations required to evaluate the convective heat transfer coefficient of a turbulent flow in a spiral heat exchanger:

$$h = \left(1 + 3.54 \frac{D_{hy}}{D_{out}}\right) 0.023 c_p \nu_m Re^{(-0.2)} Pr^{(2/3)} \quad (3)$$

where the hydraulic diameter of the channels is

$$D_{hy} = \frac{2Lb}{(L + b)} \quad (4)$$

The external diameter of the recuperator is obtained using the geometrical correlation defined by the following expression:

$$D_{out} = \sqrt{1.28L(b_{hot} + b_c + 2s) + D_{in}^2} \quad (5)$$

The inner diameter D_{in} of the spiral heat exchanger is constrained by the dimensions of the Stirling hot head and the

combustion chamber; because the diameter of the engine's head is 0.13 m and the recuperator must be placed directly over it, D_{in} was set to 0.12 m, which represents a good compromise between fitting the size of the combustion chamber and limiting the external diameter D_{out} . The thickness s of the AISI 347 foil used to separate the channels is 0.2 mm, and its thermal conduction λ is 14.2 W/mK.

The convective heat coefficients calculated by (3) can be used to define the total heat transfer coefficient of the recuperator as follows:

$$U = \frac{1}{\frac{1}{h_{hot}} + \frac{s}{\lambda} + \frac{1}{h_c}} \quad (6)$$

The required heat transfer area is

$$A = \frac{Q}{\Delta T_{lm} U F} \quad (7)$$

F is the correction factor, and ΔT_{lm} is the logarithmic mean temperature difference of the gases flowing in the channels. The correction factor can be calculated by evaluating the criterion number CN, defined as

$$CN = 2\sqrt{NTU_c NTU_{hot}} \sqrt{\pi \frac{A_c}{A}} \quad (8)$$

where NTU_c and NTU_{hot} represent the number of transfer units for the cold and hot channels. The correction factor F is

$$F = \frac{\ln(1 + CN^2)}{CN^2} \quad (9)$$

The length of the channels can be calculated from the total heat transfer area A (7):

$$L = \frac{A}{2H} \quad (10)$$

After a few iterations, the geometry of the recuperator is determined. The hot and cold channels were both set to 5 mm for the simplicity of construction; of course, this choice causes major

pressure drops in the hot channel, where the magnitude of the exhaust velocity is higher. After setting the geometry of the heat exchanger, the number of turns in the spiral can be calculated as follows:

$$n = \frac{\left[\frac{(D_{\text{out}} - D_{\text{in}})}{2} - 2s \right]}{(b_{\text{hot}} + b_{\text{c}} + s)} \quad (11)$$

To evaluate the heat transfer properties of the recuperator, the physical properties of the fresh air and the exhaust (mainly density, viscosity, thermal conductivity and heat capacity) must be defined. In this work, all of the required properties of the fluids were calculated using the NIST Refprop software program [43].

Table 2 reports the results of the iterative calculus that defines the geometry of the heat exchanger with an effectiveness of 83%.

Minton also reported a formulation to evaluate the pressure losses in the channels of the spiral plate heat exchanger:

$$\Delta P = 0.001 \frac{L}{\rho_r} \left(\frac{\dot{m}}{bH} \right)^2 \left[\frac{1.3\mu^{1/3}}{(b + 0.125)} \left(\frac{H}{\dot{m}} \right)^{1/3} + 1.5 + \frac{16}{L} \right]. \quad (12)$$

Using this expression it was possible to assess a sensibility analysis on the heat exchange performance of the heat exchanger as a function of the pressure drop in the channels. Considering the aforementioned constraints in the maximum pre-heated air temperature to avoid auto-ignition issues, with an effectiveness of 83% the pressure drop in the cold and in the hot channel is 260 Pa and 320 Pa. These values are very limited and do not affect significantly the operation of the system. Table 3 reports the trend of the pressure losses with reduced values of the heat exchanger effectiveness: the influence of a reduction of the spiral length determines limited lowering of the pressure drop in the channels. Therefore, the best option is to set the geometry of the spiral heat exchanger to its maximum possible length.

This preliminary design of the recuperator was used to numerically simulate the heat transfer performance using the commercial CFD software program FLUENT.

2.2. CFD analysis of the recuperator

CFD analysis allows to further validate the reported design procedure. CFD tool has already been successfully used to evaluate the performance of spiral plate heat exchangers in several works. In particular, the studies reported in Ref. [44] and in Ref. [39] refer to a very similar operating condition and heat exchanger design. The authors investigated the heat transfer performance of a spiral plate heat exchanger used as a recuperator for micro gas turbines; in both cases they used a 2D numerical grid and applied the $k-\epsilon$ turbulence model in the channels to solve the Navier–Stokes

Table 3

Sensibility of the pressure drop and the design parameters of the recuperator with the heat exchange effectiveness.

Pressure loss	Recuperator effectiveness				
	83%	75%	70%	65%	60%
Fresh air channel pressure loss [Pa]	260	227	212	201	193
Exhaust channel pressure loss [Pa]	320	315	312	311	310
Spiral length [m]	2.01	1.26	0.99	0.88	0.76
Number of turns	4.01	2.73	2.22	2.00	1.75

equations. Simulations have been compared to experimental data: results confirmed the capability of the CFD tool to evaluate both the heat exchange and the pressure drop in the channels being the error smaller than 10% with respect to the experimental data. The use of a simplified 2D geometry with respect to a more accurate 3D simulation only determines a slight overestimation of the velocity magnitude in the channels but does not affect significantly the evaluation of heat transfer process.

Given the very good outcomes of the abovementioned studies applied to very similar geometries and working conditions, a similar numerical analysis was also carried out in this work to validate the analytical design procedure of the 4-turn recuperator described in the previous paragraph. Fig. 2 shows a sketch of the recuperator to be coupled with the Stirling engine. Fresh air enters the spiral from the left external channel, while the exhaust enters from the right internal channel. Heated fresh air exits the recuperator from the inner left side of the spiral and can be easily conveyed to the combustion chamber.

This geometry makes the management of the flows quite simple because the warm side of the flows is closer to the hot end of the engine and to the combustion chamber, while the cold side of the flows is in the external part of the spiral.

The numerical grid is composed by a total of 1,150,692 triangular elements in a 2D geometry: about 550,000 elements are used for each of the two channels, while the remaining 50,000 elements are evenly distributed in the solid material of the spiral plate. The boundary conditions are fixed in the two inlet channels of the heat exchanger: the exhaust enters the recuperator at a temperature of 925 K, while fresh air enters at 288 K. With the mass flow rates of the two flows known, the velocity boundary condition is calculated according to the density of the fluids and channels' section. The external surface of the recuperator is considered to be adiabatic, which means that thermal losses are neglected.

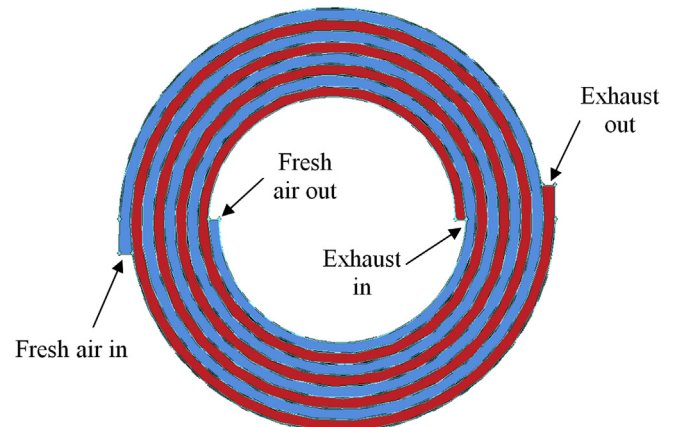


Fig. 2. Scheme of the recuperator used to recover heat from Stirling engine exhaust.

Table 2

Geometric and thermal exchange characteristics of the recuperator.

Property	Value
Exhaust channel width b_h [mm]	5
Fresh air channel width b_c [mm]	5
Spiral height H [mm]	160
Spiral inner diameter D_{in} [mm]	120
Spiral outer diameter D_{out} [mm]	202.5
Exhaust gases mean velocity $v_{m,\text{hot}}$ [m/s]	8.45
Fresh air mean velocity $v_{m,\text{c}}$ [m/s]	6.14
Wall thickness s [mm]	0.2
Total channel length L [m]	1.96
Heat transfer area A [m ²]	0.625
Total heat transfer coefficient U [W/(m ² K)]	22.326
Spiral number of turns n	4.01

Given these working conditions, a steady-state simulation was performed to define the temperature of the exhaust and the temperature of the regenerated fresh air when the system operates under nominal conditions. A realisable $k-\epsilon$ turbulent model was implemented using the enhanced wall treatment option.

The simulation convergence criteria of all of the residues were set to 10^{-6} , except that for the energy equation, which was set to 10^{-9} .

The results of the simulation confirm the validity of the preliminary design of the recuperator for a biogas volume composition of 65% methane and 35% CO₂. The relations for a turbulent flow in a spiral heat exchanger can be successfully used to design a recuperator for a biogas-fuelled burner. The spiral heat exchanger is able to pre-heat fresh combustion air and reduce the amount of fuel required to feed the Stirling engine.

Fig. 3 shows the temperature contour of the heat exchanger for both fresh air and the exhaust. The average temperature at the outlet of the fresh air channel is 541 °C, which corresponds to the design value, while the exhaust exits the heat exchanger at 196 °C. The pre-heated air can then be mixed with the fuel; the required flow of fuel is reduced to 0.225 g/s, with a significant gain in the electrical efficiency of the system. The volumetric concentration of the methane in the regenerated air is 7.5% v/v, which still falls within the flammability limit of the fuel.

The total pressure loss over the two channels is limited. As anticipated, the hot channel shows a higher pressure drop (317 Pa) than the cold one (267 Pa), as shown in Table 4.

Because the composition of the biogas can vary according to the biomass and the process that is used for its production, a sensitivity analysis with respect to the methane content of the mixture was carried out. Two more blends of biogas were simulated: biogas with a methane content of 55% and one with a methane content of 75%. The results show that there are only slight differences between the two systems, and the performance of the recuperator remains nearly constant. When the mixture contains a higher amount of methane, the required fuel mass flow rate is lower; thus, the flow rate in the hot channel is lower, while the fresh air flow rate remains constant. This obviously entails a slight reduction both of the pressure drop in the channel and of the heat transfer coefficient as also obtainable by Equation (3). The opposite behaviour is recorded when the mixture contains less methane.

Table 4

Simulation results of the regenerative heat exchanger.

Property	Value		
	65/35 methane/ CO ₂ mixture	55/45 methane/ CO ₂ mixture	75/25 methane/ CO ₂ mixture
Fresh air inlet temperature [°C]	15	15	15
Regenerated fresh air temperature [°C]	541	547	537
Exhaust temperature [°C]	650	650	650
Cooled exhaust temperature [°C]	196	203	190
Fresh air channel pressure loss [Pa]	267	270	264
Exhaust channel pressure loss [Pa]	317	328	303
Exchanged thermal power [W]	1851.7	1870.2	1834.2
Electric efficiency with the recuperator [%]	22.5	22.6	22.4

From the simulations, it can be appreciated how the difference in the performances of the three studied cases is negligible (Table 4). The average temperature at the outlet of the pre-heated fresh air channel are 547 °C and 537 °C for the 55% and 75% methane content biofuels.

As anticipated, the introduction of the recuperator reduces the amount of fuel required to feed the engine; thus, the electric efficiency of the Stirling cogenerator can be raised to up to 22.5%.

2.3. Energy and exergy performance of the system

Table 5 shows the main performance parameters of the Stirling device under study both in regenerative and non-regenerative operation modes, for a biogas methane content of 65%. It is possible to see how, in the regenerative mode, the electrical efficiency gain is significant (from 15.8% to 22.5%) while the thermal efficiency is obviously penalized; as regards the global efficiency, its value is slightly improved and reaches 90%.

In order to better assess the performance of the cogeneration device in both the regenerative and non-regenerative operation

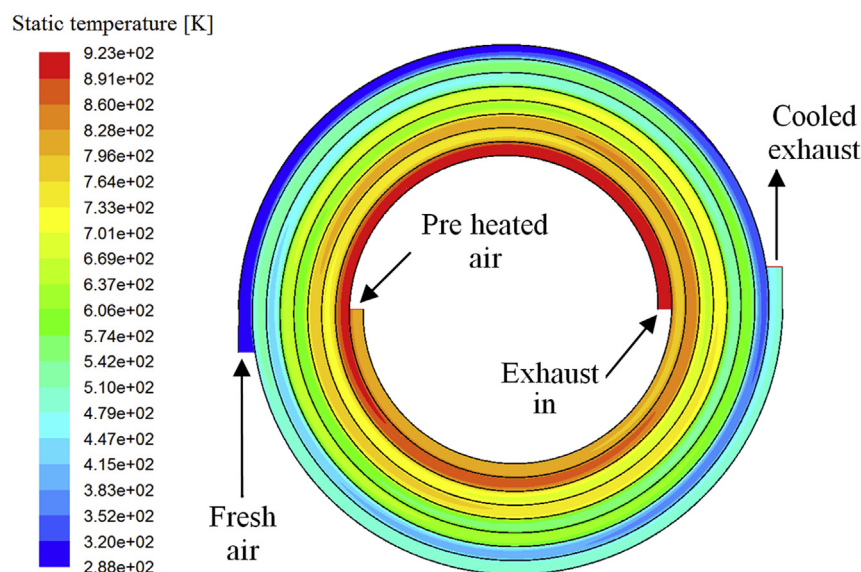


Fig. 3. Temperature contour in K over the recuperator's channels.

Table 5
Stirling cogenerator performance with a fuel methane content of 65%.

	Non-regenerative mode	Regenerative mode
Engine burner thermal power [W]	6300	4444
Stirling engine nominal electrical power [W]	1000	1000
Recuperator efficiency [%]	—	83%
Electrical efficiency [%]	15.8%	22.5%
Stirling engine cooling water thermal power [W]	3000	3000
Stirling engine exhaust thermal power [W]	1500	0
Thermal efficiency [%]	71.4%	67.5%
Global efficiency [%]	87.2%	90.0%
Power to heat ratio	0.22	0.33

modes, also an exergy analysis of the whole system has been carried out. The working conditions of the cogenerator in the hot and in the lower temperature source do not vary significantly when the recuperator is active. On the other hand, the power to heat ratio of the system has a significant change as well as the required fuel input. The exergy efficiency of the system, η_{exe} , can be evaluated using the following formulation:

$$\eta_{\text{exe}} = \frac{P_{\text{el}} + P_{\text{th}} \left(1 - \frac{T_0}{T_c}\right)}{\dot{m}_f \cdot \text{exe}_f} \quad (13)$$

where T_0 is the reference temperature (15 °C) and T_c the cold source temperature (which is set to 55 °C that represents the hot water temperature demand of the thermal user). In non-regenerative and regenerative mode the electric power output of the cogenerator is the same, while the thermal power output and the fuel mass flow rate are higher in the non regenerative mode. In the calculation, for simplicity sakes, the chemical exergy of the fuel, exe_f , has been considered equal to its LHV. The exergy efficiency in the non-regenerative mode is 24.1% while in the regenerative mode it reaches the value of 30.1%, thanks to the reduction of the fuel required to run the system.

3. Algorithm for determining the optimal dispatch of the energy system

Because the energetic, economic and environmental performances of micro-CHP devices depend on several parameters, such as i) fuel tariffs, ii) the purchasing and selling price of electricity and iii) energy loads, an optimization approach must be adopted both for the sizing of the CHP devices [45] and for their operation [46]. Optimal management of these devices allows to completely exploit their potential, particularly for micro-grid applications [47].

Fig. 4 shows the configuration considered: the Stirling engine is connected in parallel to the grid, meaning that the electric energy required by the end user can be either bought from the grid or produced by the micro-CHP device; the excess electric energy produced by the system can be sold to the grid. The micro-CHP unit is connected to a thermal energy storage (TES) unit to temporarily decouple the electricity production from heat production.

The algorithm developed, which aims at minimising operating costs, determines whether the Stirling engine should operate in the regenerative or non-regenerative mode and it identifies the optimal operation strategy of the system, including i) the electric power to be sold to the grid, ii) the electrical power to be sent to the end user, iii) the thermal power to be stored by the TES unit and iv) the thermal power directly conveyed to the end user.

The procedure requires the introduction of i) the energy load profile of the end user, which is derived using a demand forecasting tool based on historical and operation data, ii) the energy tariffs and iii) the operating and maintenance costs of the micro-CHP device. Following the minimisation of the operating cost criteria, the controller sends signals to all of the subsystems defining the optimal operation of the entire cogeneration system.

It is worthy of note that, since the performance of the Stirling unit has been defined on the basis of its efficiencies, the objective functions as well as constraints are linear, the algorithm developed turns out to be linear, taking advantage of a reduced calculation time.

3.1. Objective function

The objective function, Equation (14), minimises the operating cost of the micro-CHP systems, given by the annual sum (assessed

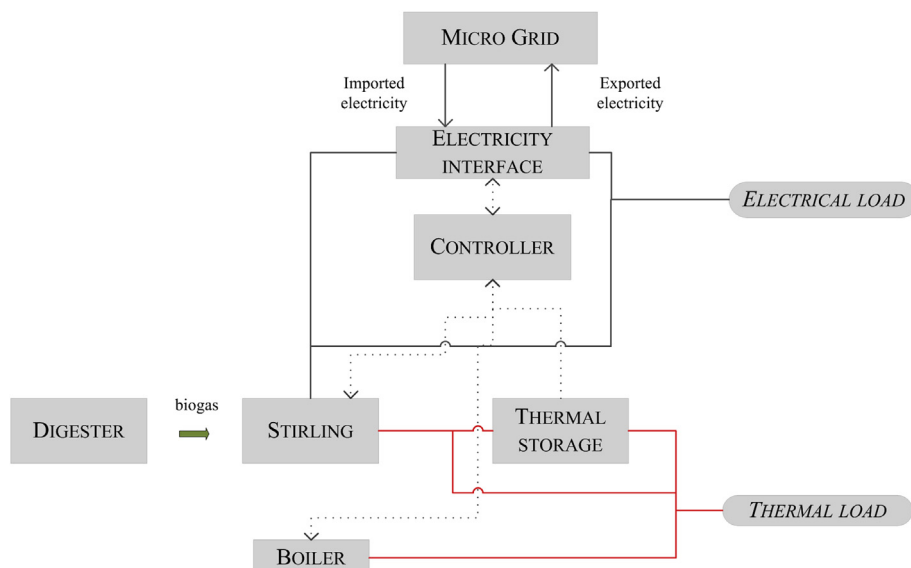


Fig. 4. Configuration and control of the Stirling cogenerator.

based on 12 typical days of operation) of i) the fuel cost of running the micro-CHP unit, given by the specific biogas cost, $c_{\mu\text{CHP}}^{\text{fuel}}$, multiplied by the amount of micro-CHP fuel consumed within a given hour, $(f_{\mu\text{CHP}})_h$; ii) the fuel cost of feeding the heating boiler, given by the specific cost of fuel, $c_{\text{boiler}}^{\text{fuel}}$, multiplied by the amount of boiler fuel consumed within a given hour, $(f_{\text{boiler}})_h$; iii) the operating and maintenance cost of the micro-CHP unit, given by the specific micro-CHP operative cost, $c_{\mu\text{CHP}}^{\text{O\&M}}$, multiplied by the electric energy produced by the micro-CHP within a given hour, $(e_{\mu\text{CHP}})_h$ and iv) the cost of purchasing electric energy from the grid, $c_{\text{el}}^{\text{buy}}$, given by the specific purchasing price multiplied by the electric energy bought from the grid within a given hour, $(e_{\mu\text{CHP}})_h$. From this amount, it is necessary to subtract i) revenues from the selling of the electric energy produced by the micro-CHP unit, given by the specific selling price, e_{sell} , multiplied by the electric energy sold to the grid within a given hour, $(c_{\text{el}}^{\text{sell}})_h$ and ii) revenues from the feed-in tariff mechanism supporting biogas energy production, $r_{\text{el}}^{\text{feed-in}}$.

$$C_{\text{Op}} = \sum_{k=1}^{12} n_k \sum_{h=1}^{24} \left[c_{\mu\text{CHP}}^{\text{fuel}} \cdot (f_{\mu\text{CHP}})_h + c_{\text{boiler}}^{\text{fuel}} \cdot (f_{\text{boiler}})_h + c_{\mu\text{CHP}}^{\text{O\&M}} \cdot (e_{\mu\text{CHP}})_h + c_{\text{el}}^{\text{buy}} \cdot (e_{\text{buy}})_h - (c_{\text{el}}^{\text{sell}})_h \cdot e_{\text{sell}} - r_{\text{el}}^{\text{feed-in}} \cdot (e_{\mu\text{CHP}})_h \right] \quad (14)$$

3.2. Main constraints

An electric and thermal balance must be achieved in each time step. The system performances were defined based on the unit's electric efficiency and its power-to-heat ratio, which for the system under analysis, varies between the regenerative and non-regenerative modes. For each time step, the procedure determines whether or not to convey the hot gases to the recuperator based on the minimum cost.

Equation (15) constrains the micro-CHP unit to produce less than its maximum installed capacity.

$$\forall h, (e_{\mu\text{CHP}})_h \leq \text{Cap}_{\mu\text{CHP}} \quad (15)$$

Finally, Eqs. (16) and (17) define the performance of the TES unit. The first equation states that the total amount of heat stored at the beginning of an hour is equal to the heat stored in the previous hour plus the heat stored in that hour minus the heat released to the end user in that time step. The second equation prevents the heat stored from exceeding the TES capacity.

$$\forall h, (\text{heat}_{\text{TES}})_h = (\text{heat}_{\text{TES}})_{h-1} + (\text{heat}_{\text{TESin}})_h - (\text{heat}_{\text{TESout}})_h \quad (16)$$

$$\forall h, (\text{heat}_{\text{TES}})_h \leq \text{Cap}_{\text{TES}} \quad (17)$$

During the optimisation, some assumptions are made:

- the micro-CHP system can operate from zero to the maximum capacity
- the operating cost of the TES unit is neglected
- the TES unit has a capacity of 6 kWh_{th}
- micro-CHP performance data are defined based on the electrical efficiency and the power-to-heat ratio, which in this case, are 0.22 in the non-regenerative mode and 0.33 in the regenerative mode.

4. Case study

4.1. Energy loads and main parameters

A single dwelling was considered in the case study. Energy demand data were estimated hourly for an average day of each month, according to the procedure suggested in Ref. [48]. The inputs of the model are end-user type, geographic location, maximum thermal power in winter, maximum cooling power in summer, maximum thermal power for hot water and maximum electric power.

The electric load was assumed to be independent of the season, while the thermal load was variable. Fig. 5 shows the thermal demand for a typical day in winter (January), fall/spring (April) and summer (July) for an apartment located in central Italy and the hot water demand, which has been considered to be independent of the seasons.

The parameters used in the analysis are shown in Table 6. For the reference case, the cost of biogas was assumed to be 0.03 €/kWh, resulting in a low biomass heating value of 6 GJ/ton [32], and biomass cost of 50 €/ton. To assess the influence of these values on the model's results, a sensitivity analysis was conducted.

The situations investigated were: i) the cost of feeding the boiler is higher than that of feeding the micro-CHP unit because, in the case of low biogas availability, natural gas could be used to feed the additional heating boiler; ii) the cost of biogas is zero, in the case of biomass availability; and iii) a feed-in tariff for biogas electricity production of 0.18 €/kWh, which is characteristic of the Italian legislative framework [49]. The last case was considered since several countries such as Germany, France, Croatia, Greece and UK [50] provide a similar support mechanism to incentivise the production from biogas.

4.2. Results and discussions

Figs. 6 and 7 show the main operating parameters of the micro-CHP unit for a typical winter day (typical day in January) and fall/spring day (April). The system operates at its maximum capacity and always in the regenerative mode, as indicated in Table 6, due to the higher incomes derived from a better electrical efficiency of the system.

The additional heating boiler satisfies the heat demand not covered by the Stirling engine, and the electricity not produced by the unit is bought from the grid.

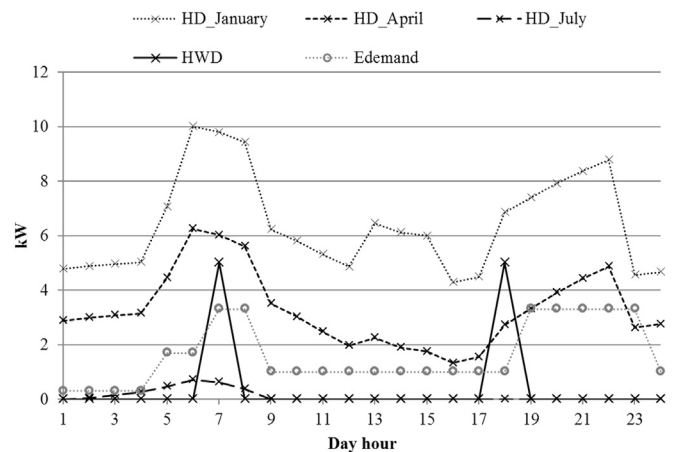


Fig. 5. Energy loads of the single dwelling under analysis.

Table 6
Parameters assumed in the analysis for the reference case.

Parameters	Reference case	Case 1	Case 2	Case 3
Cost of biogas feeding the micro-CHP unit [€/kWh]	0.030	0.030	0	0.030
Cost of biogas feeding the boiler [€/kWh]	0.030	0.035	0	0.030
Electricity purchasing price [€/kWh]	0.150			
Electricity selling price [€/kWh]	0.090			
O&M costs [€/kWh]	0.018			
Feed-in tariff [€/kWh]	0	0	0	0.180
Cost for the high-efficiency recuperator [€]	200			

In summer, the system operates following the electrical demand, and the Stirling co-generator is switched off when the capacity of the TES unit is reached (Fig. 8).

To assess the validity of the solution proposed in this study, Table 7 shows the main technical and economic performance data obtained from the simulation of the proposed system and the standard Stirling engine device, both compared with separate energy production. It can be concluded that the system proposed provides an increase in electricity production of approximately 38%, with a consequent increase in the number of operating hours and savings. As expected, both solutions show interesting energy and economic advantages compared to the standard system device: CO₂ emission is reduced by 6% in the standard case with an economic gain of €400; in regenerative case CO₂ emission lowers by 9% with an economic gain of €600. Comparing the proposed system with the standard Stirling device, the yearly gain is approximately €150; considering a cost of about €200 for the high-efficiency recuperator, the pay-back period of the recuperator is less than two year.

Finally, Table 8 shows the results of the sensitivity analysis, whose parameters were discussed in section 4.1 (Table 6).

The difference in the cost of biogas for boiler feeding (case 1) makes the non-regenerative mode adequate in minimising the operating costs in the case of high thermal demand. A null cost of biogas (case 2), as expected, increases the economic and energy gains derived from micro-CHP production: in this case, the system operated in the non-regenerative mode for 214 h, but it is worth noting that there was no difference in the total number of operating hours of the micro-CHP unit.

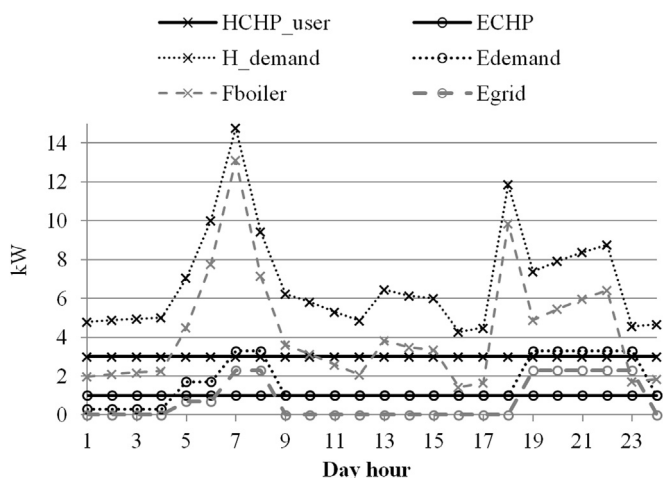


Fig. 6. Main simulation results for a typical winter day (January).

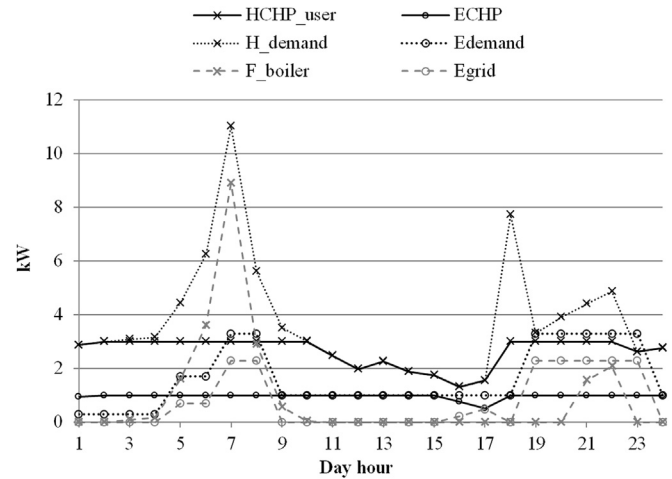


Fig. 7. Main simulation results for a fall/spring day (April).

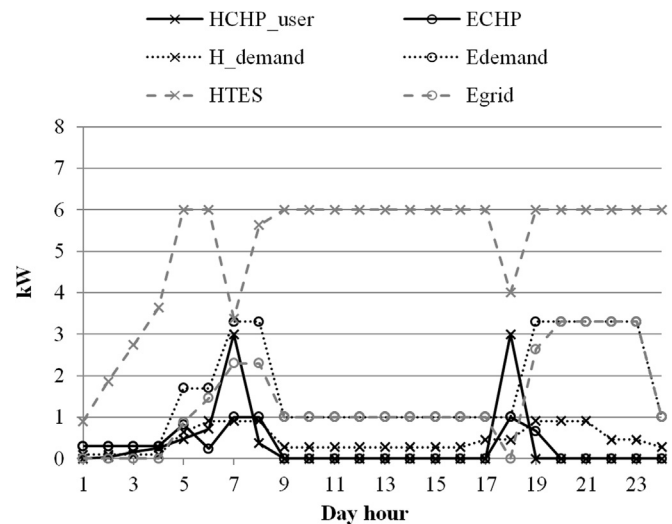


Fig. 8. Main simulation results for a summer day (July).

In case 3, in which there are revenues from the feed-in tariff mechanism, the system always operates in the regenerative mode in order to take advantage of the feed-in tariff on the electricity produced. Clearly, there is a consistent increase in revenues, suggesting the need to introduce support mechanisms for the spread of renewable electricity generation.

Table 7

Results of one-year analysis of the system under study compared to those of the standard Stirling engine device (reference case).

Parameters	Solution under study	Standard Stirling engine device
Electricity produced by micro-CHP [kWh]	5530	4001
Fuel feeding the heating boiler [kWh]	8099	5065
Electricity sold to the grid [kWh]	502	324
Energy bill of separate production	€2871	€2871
Energy bill of the solution micro-CHP	€2300	€2450
CO ₂ separate production [tonCO ₂ /year]	14.4	14.4
CO ₂ solution micro-CHP [tonCO ₂ /year]	13.3	13.5
N° operating hours in regenerative mode	5428	0
N° operating hours in not regenerative mode	0	4052

Table 8
Sensitivity analysis.

Parameters	Reference case	Case 1	Case 2	Case 3
Electricity produced by micro-CHP [kWh]	5530	5670	5656	5670
Fuel feeding the heating boiler [kWh]	8099	5077	7338	7655
Electricity sold to the grid [kWh]	502	831	817	831
Energy bill of separate production	€2871	€2996	€2119	€2871
Energy bill of the solution micro-CHP	€2300	€1319	€301	€2300
CO ₂ separate production [tonCO ₂ /year]	14.4	14.4	14.4	14.4
CO ₂ solution micro-CHP [tonCO ₂ /year]	13.3	13.3	13.2	13.2
N° operating hours in regenerative mode	5428	4123	5456	5670
N° operating hours in not regenerative mode	0	1547	214	0

5. Conclusions

The present work aimed to assess the potential of applying a high-efficiency recuperator to a 1 kW_e Stirling engine device to increase its electrical efficiency. An optimal energy management algorithm able to optimise the operation of a micro-CHP system was developed and applied to a residential case study to assess the potential of this system. The extra cost of the high-efficiency recuperator can be easily recovered in less than two years due to the gains derived from a higher electrical efficiency, helping also the manufacturer to increase the market in those areas characterised by long period of low heat demand, such as all Mediterranean countries. Furthermore the increase in the electrical efficiency of the Stirling device, from 15.8% to 22.5%, can reinforce the spread of local biomass resources which, in several urban areas in both industrialised [51] and non-industrialised countries [52], appear to offer a real opportunity. The implementation of such a solution can incentivise the application of Stirling devices in micro-grid applications providing extra environmental and economic gains of approximately 6% and 35% if compared to the standard Stirling device. Finally, since its production can be easily programmed, the proposed system can be applied in combination with other variable renewable energy devices without the need to adopt expensive energy storage units and, thus, it is possible to rely on only renewable sources.

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Nomenclature

A	heat transfer area, m ²
A _c	channel section, m ²
AF	air/fuel ratio
b	channel width, m
C	cost, €
c	specific cost (€/kWh)
Cap	capacity, kW
c _p	specific heat, J/(kg K)
D	diameter, m
ΔT _{lm}	logarithmic mean temperature difference, K

e	electric energy, kWh
ex	combustion air excess
exe	chemical exergy of the fuel
f	fuel consumption, kWh
F	correction factor
h	heat transfer coefficient, W/(m ² K)
H	spiral height, m
η	efficiency
heat	heat, kWh
L	spiral length, m
LHV	low heat value, J/kg
$\dot{m}$	fresh air mass flow rate, kg/s
μ	viscosity, kg/(m s)
μCHP	micro-combined heat power system
NTU	number of transfer units, 1/s
P	power, W
Pr	Prandtl number
r	specific revenues, €/kWh
Re	Reynolds number
s	spiral thickness, m
T	temperature, °C
U	global heat transfer coefficient, W/(m ² K)
v _m	mean velocity, m/s
λ	thermal conductivity, W/(m K)

Subscripts and superscripts

b	burner
c	cold
comb	combustion
el	electric
exe	exergetic
f	fuel
h	hour
hot	hot
hy	hydraulic
in	inner
O&M	operating and maintenance
op	operating
out	outer
TES	thermal energy storage
th	thermal
0	reference condition

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